Effect of Variation of Initial and Back Pressure On the Economy of a Simple, High Speed, Automatic Steam Engine

A. H. Boehmer
J. B. Wells

1907

621.17
B 63

ARMOUR
INST. OF TECH. LIB.
CHICAGO.

Illinois Institute
of Technology
Libraries

AT 61

Boehmer, Alexander Henry
Effect of the variation of
initial and back pressure

EFFECT OF THE VARIATION OF INITIAL AND BACK PRESSURE ON THE ECONOMY OF A SIMPLE, HIGH SPEED, AUTOMATIC STEAM ENGINE

A THESIS

PRESENTED BY

Alexander Henry Boehmer

John Barnard Wells

TO THE

PRESIDENT AND FACULTY

OF

ARMOUR INSTITUTE OF TECHNOLOGY

FOR THE DEGREE OF

BACHELOR OF SCIENCE IN MECHANICAL ENGINEERING

HAVING COMPLETED THE PRESCRIBED COURSE OF STUDY IN

MECHANICAL ENGINEERING

JUNE 1907

ILLINOIS INSTITUTE OF TECHNOLOGY
PAUL V. GALVIN LIBRARY
35 WEST 33RD STREET
CHICAGO IL 60616

[illegible]

EFFECT OF THE VARIATION OF INITIAL AND BACK PRESSURE ON THE ECONOMY OF A SIMPLE, HIGH SPEED, AUTOMATIC STEAM ENGINE.

The performance of a steam engine is often expressed in terms of a certain weight of water per horse-power per hour. This is a very indefinite rating for comparison except where conditions of initial and back pressure and quality are the same, as the weight of steam used is largely dependent on these conditions. This series of tests was undertaken for the purpose of determining what relation exists between the economy of a steam engine and the initial and nack pressures.

The engine used in these tests was a Phoenix engine with cylinder dimensions 8 1/2" X 10 ", rated at 40 horse-power, and located in the steam laboratory of The Armour Institute of Technology. It is a high speed, automatic engine of the piston valve type with an enclosed crank case. The valve is of the outside admission type and takes its motion, through a rocker arm, from a Rites inertia governor in the fly wheel as shown in the photographs. There are four pistons, the inner two

19491

being merely for the purpose of increasing the bearing surface and thus decreasing the wear. A long stuffing box is provided in the cylinder head and hand holes are left in the sides of the case to facilitate packing it. Near this stuffing box is a diaphragm with a small stuffing box which keeps leakage from the cylinder out of the crank case and prevents the crank case lubrication from being splashed out through the hand holes. The cross head is vertical and works in bored guides. The wrist pin end of the connecting rod is solid and is fitted with bronze boxes adjusted by a wedge and screws. The crank pin end of the connecting rod is of the marine type, having babbitted steel boxes with holes drilled in the upper half to admit oil to the crank pin. A small channel near the top of the hood covering the crank case conducts oil to the main bearings through 1/4 inch pipes provided with stop valves for regulaiting the flow. A large hand hole is placed in the left side of the case to admit of access to the wrist pin boxes and cross head slippers to adjust them. The main bearings are babbitted and are set at an angle of 45°.

The load on the engine was applied by a special type of Prony brake. The brake pulley is of the internally flanged type of 30" diameter and 12"

face and cooled by water ciraulating inside of it. The flow of water was regulated so that it evaporated as fast as it was supplied. The brake is in two sections, each being built of boards set on edge slightly separated and curved to fit the pulley. The two parts are held together by a long bolt at each end. The bolt farthest from the cylinder has a hand wheel on the top in place of a nut in order to vary the load. This bolt is slipped through a piece of 3/4" pipe extending the full length of the bolt and fitted with a tee at each end. Pipes are lead from these tees and are bent so as to join opposite the center of the brake pulley. This gives great flexibility in a horizontal plane and thus elimates much of the vibration common to other forms of Prony brakes.

Steam is supplied to the engine through a 2" pipe and exhausts through a 3 1/2 " pipe. A Crane vertical separator is placed in the supply pipe as shown in order to give as dry steam as possible to the engine. The exhaust pipe, 72 feet long, with four valves, seven 90° bends, and two 45° bends, leads to a Wheeler surface condenser. From this the exhaust steam is pumped into the weighing tanks. The throttle in the supply pipe regulates the pressure of the steam supplied to the engine.

The gauge which is tapped into the lower part of the throttle valve, as shown in the photographs, indicates the initial pressure of the steam. The back pressure is regulated by means of a valve in the exhaust pipe just below the steam chest. As shown in the photographs a gauge is tapped in just above this valve to indicate the back pressure.

A Peabody throttling calorimeter was used to determine the quality of the steam admitted to the engine. Because it was desirable to tap as few holes as possible in the valve casing the calorimeter was attached to the steam pipe above the valve The quality of the steam delivered to the engine was determined by assuming that there was no radiation between the calorimeter nipple and the valve. As may be seen in the photographs the distance between them is so short that the radiation will be negligible and the above assumption is justified. The steam from the calorimeter was discharged through a 3/4 " pipe to the bucket below the cylinder. The pressure in the calorimeter was measured by a mercury manometer graduated to quater-inches.

Two Crosby inside- spring indicators were used, one at each end, which is the method reccomended by The American Society of Mechanical Engineers.

The reducing motion is a special pattern furnished by the builders, The Phoenix Iron Works Company, with all their engines. A piece of steel rectangular in cross-section is bloted to the cross head with its crank end higher than its head end. A roller which is carried on the short arm of a bell-crank lever runs on the under side of this bar. The arms of this lever are off-set, the shaft passing through the cover of the hand hole opposite the cross head. The arms are at right angles, the longer arm, which is on the outside, being vertical at the middle point of its travel. The indicator cord is fastened to the upper end of the long arm. This motion has been tested for geometrical error which was found to be very small and can be neglected.

Two series of tests were made, one keeping the back pressure constant and varying the initial pressure, and the other keeping the initial pressure constant and varying the back pressure. The first series was run on a back pressure of 12 pounds gauge, and the initial pressure varied between 80 and 120 pounds inclusive by five pound steps. The second series was run at an initial pressure of 100 pounds gauge and the back pressure varied between 6 and 20 pounds inclusive by two pound steps. Six pounds was used as the

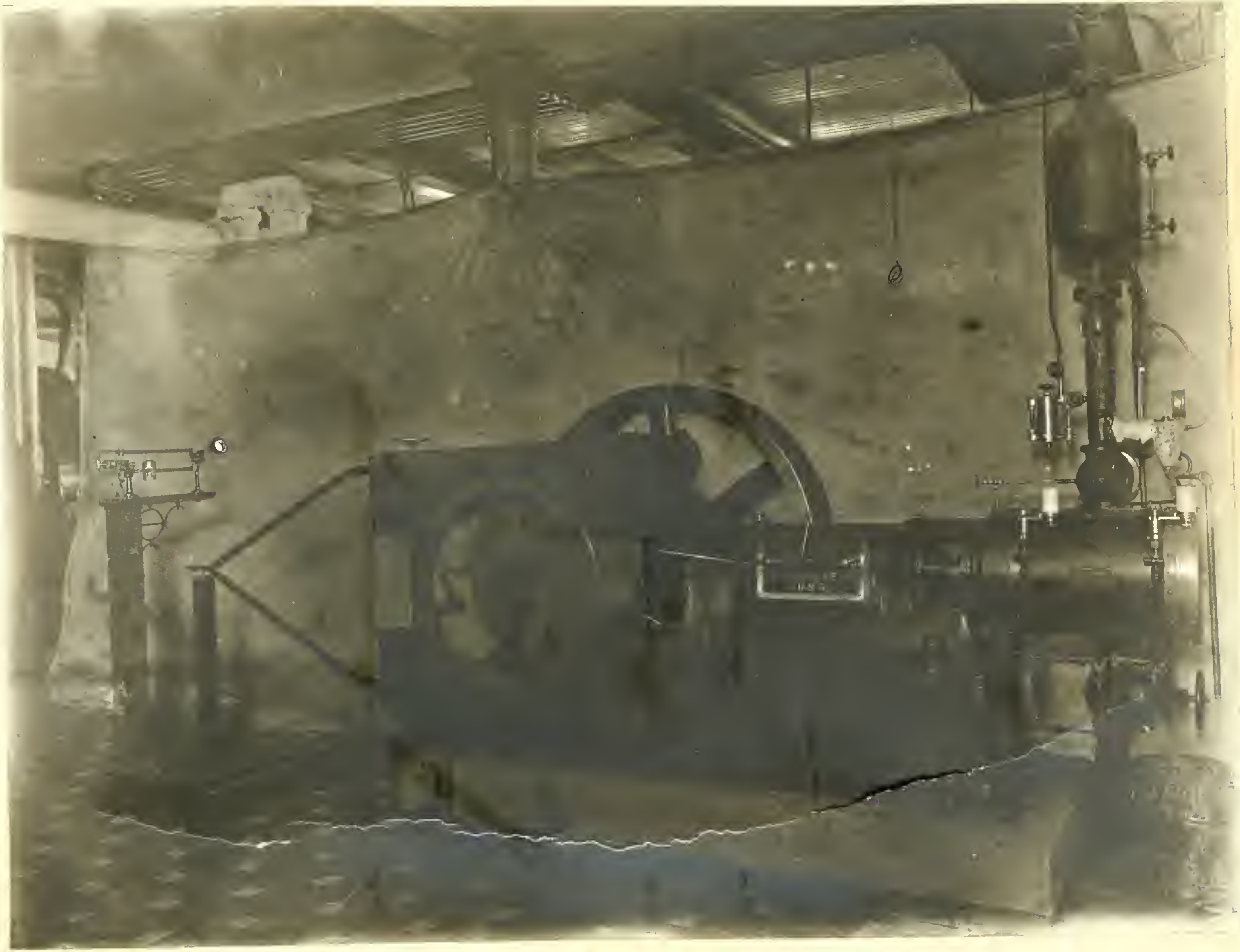
lowest back pressure as it was the best that could be obtained wtith the load used. This was due to the size and length of the exhaust pipe and the number of bends and valves. The load was kept constant for two reasons; the first being to eliminate any effect a varying load might have on the water rate, and the other being to make the conditions similar to those in commercial work where the daily load curve practically repeats itself.

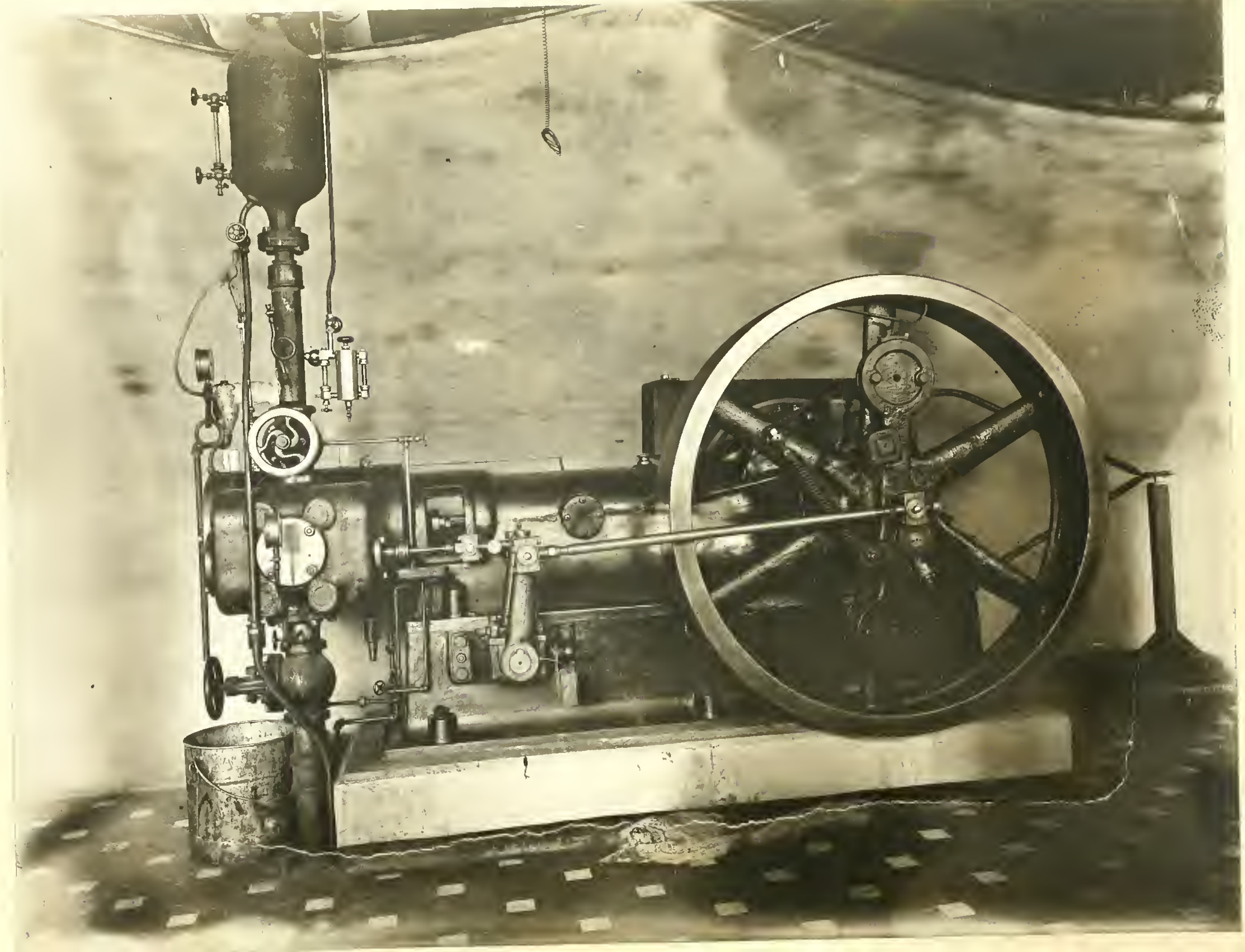
The engine was started and run for some time before beginning a test. During this time the load and pressures were adjusted in order to have all conditions constant before starting a test. Before a run was started, the separator was drained and the tube connecting the calorimeter was disconnected and the water of condensation allowed to run out. The latter percaution was taken to prevent any error due to stem condensing abov the mercury column. The runs were twenty minutes long, readings and indicator cards being taken every five minutes. After the first run the interval between tests was ten minutes. The pressures were adjusted for one run as soon as the previous run was completed.

In figuring the quality of the stea supplied to the engine the heat in one pound of steam was found

from the equation, $H = \lambda + .48(t_1 + t_2)$, where λ is the total heat in a pound of steam corresponding to the calorimeter pressure, and the $(t_1 + t_2)$ is the degree of superheat in the calorimeter. IF no radiation takes place between the steam pipe and the calorimeter, $H = x_1 r_1 + q_1$, where x_1, r_1, and q_1 are the properties of the steam at the pressure in the stem pipe. It may be assumed that no radiation occured between the steam pipe and the calorimeter as the connection was short and carefully lagged. As has been already shown the radiation between the calorimeter nipple and the throttle is so small as to be negligible. On this account the amount of heat in a pound of steam is the same below the throttle as in the steam pipe; hence $x_2 r_2 + q_2 = x_1 r_1 + q_1$, where x_2, r_2, and q_2 are the properties of the steam below the valve. Hence $H = \lambda + .48(t_1 + t_2) = x_2 r_2 + q_2$, and x_2 can be solved for directly.

The water rate was found by dividing the actual weight of water by the indicated horse-power and the brake horse-power. This value was then corrected for the quality by multiplying by the per cent of dry steam as shown by the calorimeter. The amount of heat charged against the engine per horse-power per hour is $W(x_1 r_1 + q_1 - q_4)$, where W is the weight





of wet steam per horse-power per hour, x_1 , r_1 , and q_1 are the porperties of the steam supplied and q_4 is the heat of the liquid corresponding to the exhaust pressure.

The dead load on the brake was determined by turning the engine slowly by hand and weighing the load on the scales. The engine was the n turned backward at as nearly the smae speed as possible and the load again weighed. The average of the two scale readings is the dead load to be subtracted from the scale reading to give the net load on the scale. The friction between the loose brake and the pulley makes the apparent dead load too large when the engine is turned forward and too small by an equal amount when the engine is turned backward at the same speed. Therefore the average is the correct reading.

The gauges were calibrated on a dead weight bauge tester, the readings being taken while the piston of the tester was revolving. Readings were taken ascending and descending and the average of each pair plotted against the tester reading. The true pressures given in the data were taken from the curves obtained in this way.

The indicators were calibrated against the high pressure gauge and were found to be without sensible

error. The drum springs and pencil motions were not tested.

The scales under the weighing tanks were tested and found to be correct and as the scale used was new it was assumed to be correct without investigation.

Curves were plotted for each series of tests using the variable pressures as abscissae and the water rate and thermal efficiencies as ordinates. An examination of the curves shows that the decrease in the water rate is proportional to the increase in theinitial pressure up to a pressure about one and one-quater times the initial pressure for which the engine was designed, that is, 85 or 90 pounds. After that the increase is not so rapid. It is probable that when a certain initial pressure is reached the economy would remain constant. The curves of thermal efficiency are practicall the inverse of those for the water rate.

From the curves plotted from the data obtained in the runs with variable back pressure it can be seen the the water rate increases directly with the increase in the back pressure. The curves show no tendency to depart from a straight line within the range of pressures used. As before the curves of thermal efficiency are practically the inverse of those for the water rate.

Errors effecting the results are probably small. One chance for error lies in our inability to maintain a constant back pressure at all times. The pressure ranged from about one pound below the desired pressure to the same amount above These variations were due to the fact that the exhaust was throttled and as the engine used varying amounts of steam the exhoust pressure varied also, rising with an increased amount of steam and falling with a decreased amount. The initial pressure varied through a little greater range due to the fact that other apparatus was using steam from the same boiler plant at the same time. Inasmuch as one man watched both gauges, the calorimeter, and took cards, it was impossible to attend to the valves closely enough to prevent these variations of pressure.

The speed was taken with a hand speed counter, but the error due to this cause is not over .5%.

There are four valves in the exhaust line which might have leaked to some extent as we had no means of ascertaining their tightness without disconnecting them. This could not be done for lack of time as it would have had to be done every time tests were made owing to the fact that tests were being run on two other pieces of apparatus, which delivered steam through these valves to the condenser.

Reference was made to the Code of Engine Testing as given in the Proceedings of The American Society of Mechanical Engineers, Vol. 24, pp. 713 - 846. The report on the Exhibits and Tests of the Pennsylvania R.R. at the St. Louis Exposition and the pamphlet of the Engineering Lore Series published by The Buffalo Forge Company of Buffalo, N.Y. were also consulted. This last is the report of a test conducted by Professor R.C.Carpenter of Sibley College, Cornell University on a high speed, tanden-compound engine of their make.

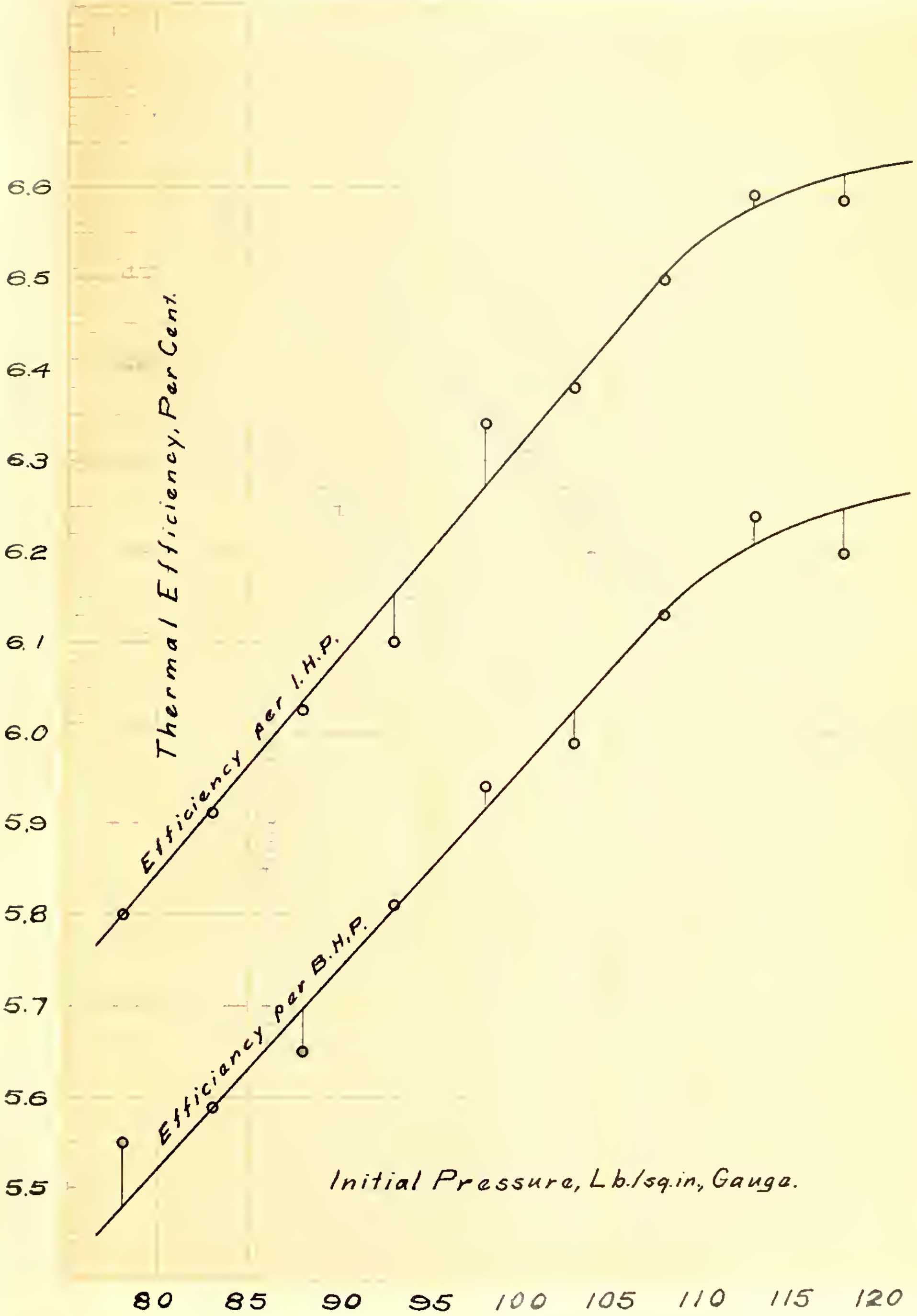
Run	R.P.M.	I.H.P.	B.H.P.	Mechanical Efficiency	% Quality
1	291	37.2	35.5	95.4	99.4
2	293	37.8	35.8	94.7	98.9
3	292	38.0	35.6	93.7	99.2
4	293	37.6	35.8	94.9	99.1
5	295.4	38.4	36.0	93.8	98.9
6	293.8	37.8	35.8	94.7	99.0
7	293.2	38.0	35.7	93.9	99.1
8	293.8	37.7	35.8	95.0	99.1
9	295	38.0	36.0	94.8	99.0
10	294	37.9	35.9	94.75	98.95
11	293.4	38.2	35.8	93.8	99.0
12	294.8	38.3	36.0	94.0	98.9
13	293.4	38.1	35.8	94.0	98.9
14	294.6	38.2	35.9	94.0	98.5
15	295.6	38.2	36.1	94.5	98.55
16	296	38.2	36.1	94.5	98.5

Variable Initial Pressure.

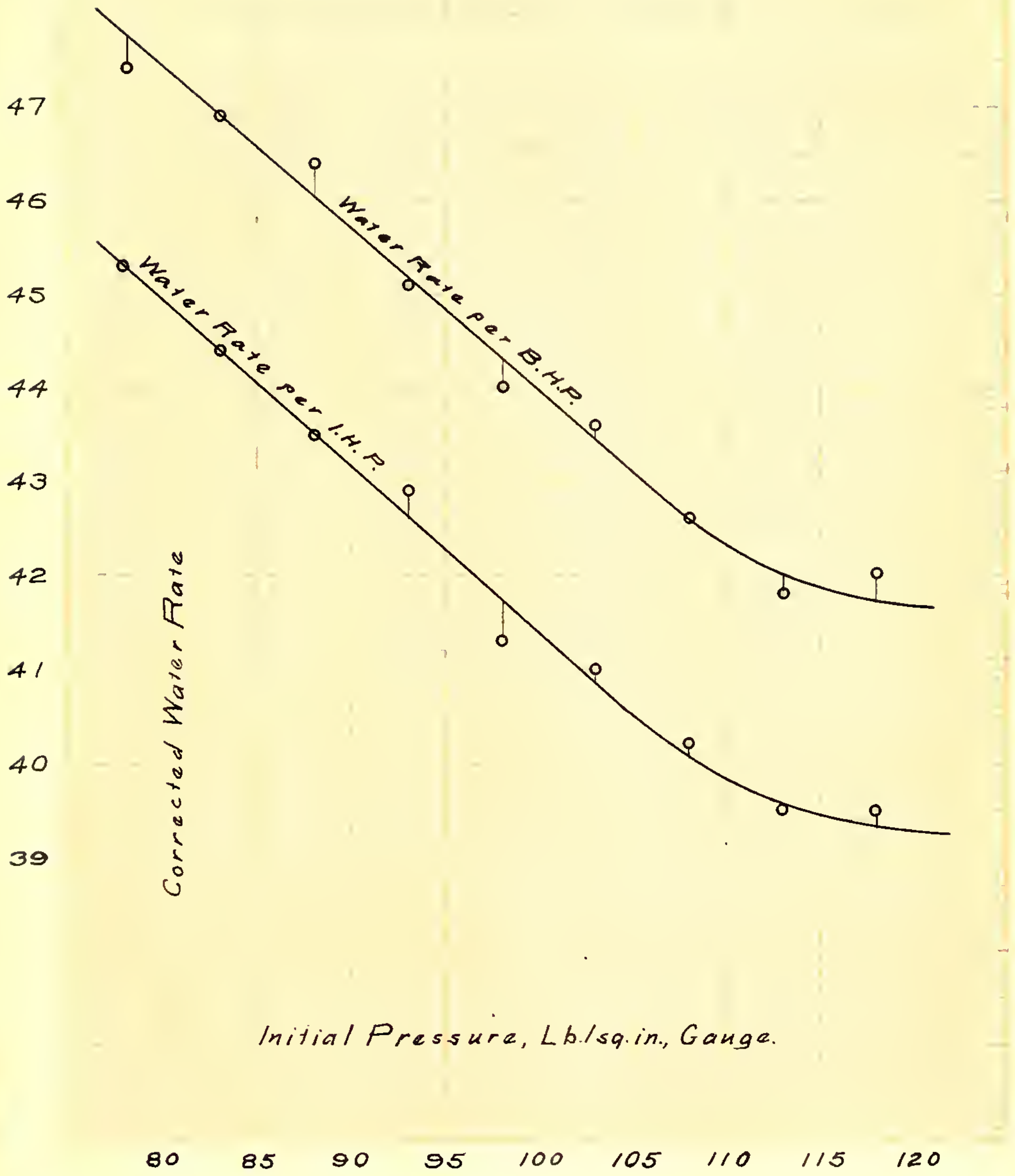
12 Pounds Constant Back Pressure.

	Initial	Water Rate		Thermodynamic Efficiency	
Run	Pressure	I.H.P.	B.H.P.	I.H.P.	B.H.P.
1	78	45.6	47.7	5.80	5.55
2	83	44.9	47.4	5.91	5.59
3	88	43.9	46.8	6.03	5.65
4	93	43.3	45.5	6.10	5.81
5	98	41.7	44.5	6.34	5.94
7	103	41.4	44.0	6.38	5.99
16	108	40.8	43.2	6.50	6.13
15	113	40.1	42.4	6.60	6.24
14	118	40.1	42.6	6.59	6.20

Curves for Thermal Efficiency with Variable Initial Pressure.



Curves for Corrected Water Rate with Variable Initial Pressure.

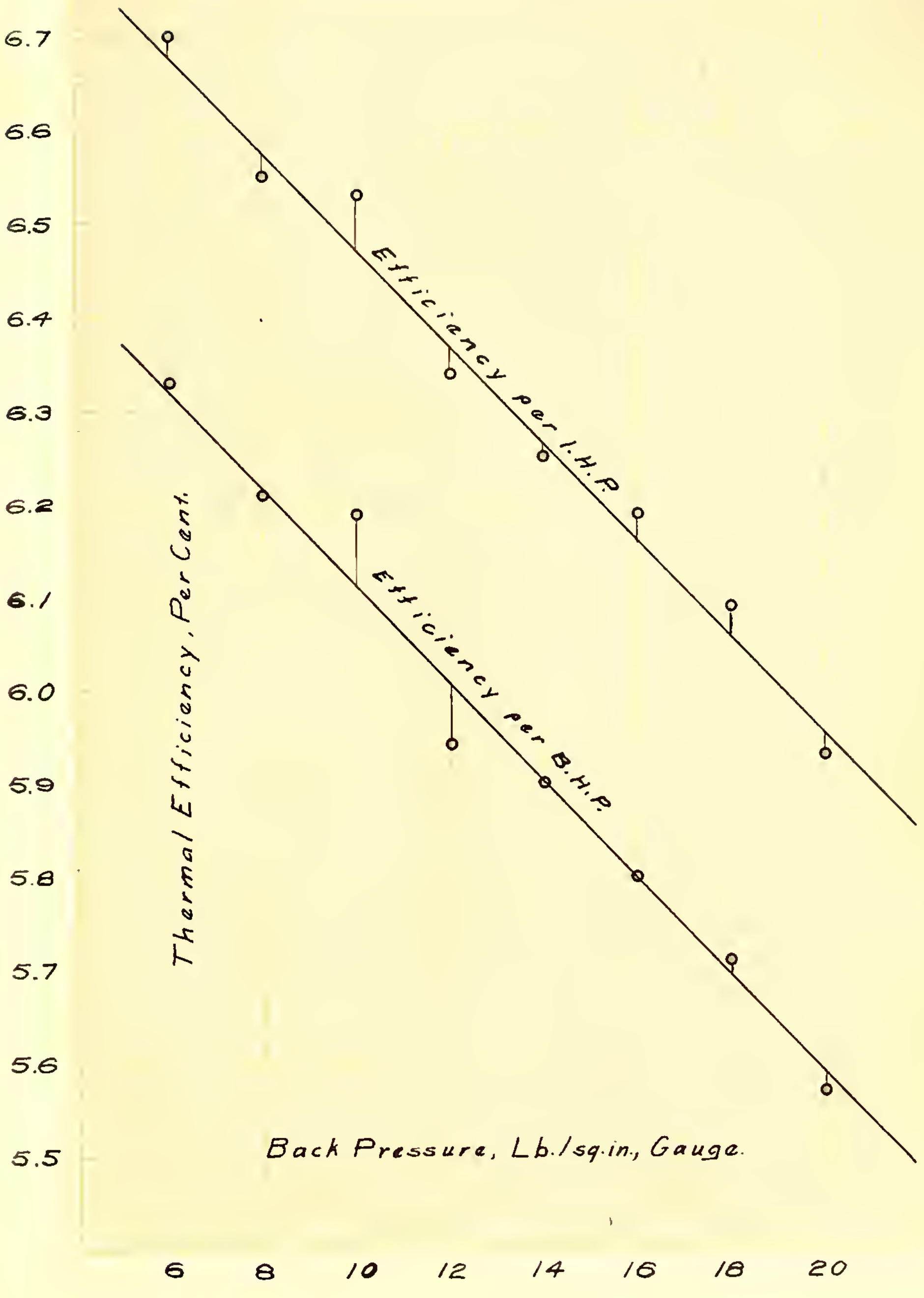


Variable Back Pressure.

98 Pounds Constant Initial Pressure.

Run	Back Pressure	Water Rate I.H.P.	Water Rate B.H.P.	Thermodynamic Efficiency I.H.P.	Thermodynamic Efficiency B.H.P.
6	6	38.9	41.1	6.70	6.33
8	8	39.9	42.1	6.55	6.21
9	10	40.3	42.5	6.53	6.19
5	12	41.7	44.5	6.34	5.94
10	14	42.3	44.7	6.25	5.90
11	16	42.8	45.7	6.19	5.80
12	18	43.8	46.6	6.09	5.71
13	20	45.1	48.0	5.95	5.57

Curves for Thermal Efficiency with Variable Back Pressure.



Curves for Corrected Water Rate with Variable Back Pressure.